Case Studies in Thermal Engineering 8 (2016) 403–413

Contents lists available at ScienceDirect

Case Studies in Thermal Engineering

journal homepage: www.elsevier.com/locate/csite

The use of parabolic trough collectors for solar cooling – A case study for Athens climate

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ARTICLE INFO

Keywords:
Parabolic trough collector
Solar cooling
Parametric analysis
Numerical model
Absorption chiller

ABSTRACT

Solar cooling is a state of the art technology in the last years, especially for countries with high irradiation level as Greece. The objective of this study is to determine the performance of an efficient solar cooling system for Athens, the most populated city in Greece. The examined solar cooling system is a one stage water/lithium-bromide absorption chiller driven by a parabolic trough collector coupled with a storage tank. A numerical model is developed in order to simulate the dynamic performance of this system. Many parameters have been investigated through sensitivity analyses and their optimum values are determined. The mass flow rates and the storage tank volume are the parameters that have been examined parametrically. The final results proved that by using a PTC module with an aperture area of 14 m^2, a building area of about 25 m^2 can be cooled for 13 h daily during the summer period. The optimum specific mass flow rate was determined to 0.03 kg/sm^2 and the optimum storage tank volume to 0.3 m^3. Moreover, a case study for a typical building of 100 m^2 is presented with very satisfying results, where four PTC modules are used in parallel connection.

1. Introduction

Nowadays, our society faces many energy problems because of the fossil fuel depletion, the increasing rate of CO_2 emissions, the increasing rate of the electricity price and the new lifestyle trend which are fully connected with high energy consumption. These problems can be easily faced by using renewable and sustainable energy sources [1]. Solar energy utilization is the most widespread method for covering a part of the thermal or electrical needs of the buildings. An application which is able to use solar energy and to reduce the electricity consumption is the solar cooling technology with sorption machines. Absorption chiller is the most mature technology among the sorption technologies [2] and the combination of this with concentrated collectors is an interesting idea which is investigated the last years. Especially for countries with high irradiation levels, the use of solar cooling technologies can be an attractive investment which is environmental friendly. In Greece the annually incident solar energy is 1600 kW h/m^2 and especially in Athens the daily average radiation is approximately 4.35 kW h/m^2 [3]. These data show that the solar potential in Greece is high enough to support the idea of solar cooling.

The use of concentrated collectors gives the opportunity to operate in high temperatures levels (> 100 °C) with great thermal efficiency, using lower collecting area than the other solar technologies. Moreover, PTC is more mature technology among concentrating thermal collectors making choice to be financial more feasible. The use of a storage tank in the examined system leads to temporary energy storage for utilization after sunset hours. The working fluid in the system is pressurized water in order to be

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http://dx.doi.org/10.1016/j.csite.2016.10.003
Received 21 September 2016; Received in revised form 10 October 2016; Accepted 27 October 2016
Available online 29 October 2016
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Nomenclature

A	Area [m^2]
C	Concentration ratio [Dimensionless]
c_p	Specific heat at constant pressure [kJ kg^{-1} K^{-1})
COP	Coefficient of performance [Dimensionless]
D	Diameter [m]
f	Focal length [m]
F_R	Heat removal factor [Dimensionless]
h	Heat transfer coefficient [Wm^{-2} K]
G_b	Beam radiation [Wm^{-2}]
K	Incident angle correction factor
k_m	Thermal conductivity [Wm^{-1} K^{-1}]
L	Tube length [m]
M	Total mass [kg]
m_c	Collector mass flow rate [kg s^{-1}]
m_L	Load mass flow rate [kg s^{-1}]
m_a	Specific mass flow rate in tube [kgs^{-1} m^2]
n	Counter [Dimensionless]
Nu	Nusselt number [Dimensionless]
n_o	Optical efficiency [Dimensionless]
t	Time [s]
T	Temperature [°C]
P	Pressure [bar]
P_r	Prandtl number [Dimensionless]
q	Cooling load [W]
q_L	Heat input in the chiller [W]
Q_{loss}	Tank thermal losses [W]
Q_{ref}	Refrigeration output [kW h]
Qu	Useful energy, [W]
Re	Reynolds number [Dimensionless]
U_L	Collector loss coefficient [Wm^{-2} K^{-1}]
U_T	Tank loss coefficient [Wm^{-2} K^{-1}]
V	Storage tank volume [l]
V_{air}	Air velocity [ms^{-1}]
W_a	Width [m]

Greek Characters

α	Receiver absorbance [Dimensionless]
γ	Intercept factor [Dimensionless]
δ	Declination angle [o]
ε	Emittance [Dimensionless]
η	PTC efficiency [Dimensionless]
θ	Solar incident angle [°]
θ_z	Zenith angle [o]
ρ	Trough reflectance [Dimensionless]
σ	Stefan-Boltzmann constant [Wm^{-2} K^{-4}]
τ	Cover transmittance [Dimensionless]
ω	Solar Hour angle [°]

Subscripts and superscripts

a	Aperture
abs	Absorbed
am	Ambient
c	Cover
ci	Cover inner
co	Cover outer
f	Fluid mean
fi	Fluid inlet
i	Inlet
o	Outlet
r	Receiver
r,c-am	Radiation between cover and ambient
ri	Receiver inner
ro	Receiver outer
r,r-c	Radiation between receiver and cover
S,max	Storage tank maximum
sky	Sky conditions
S1	Over part of the tank
S2	Medium part of the tank
S3	Down part of the tank
T	Tank

kept in liquid phase in high temperature levels.

The combination of PTC with an absorption chiller is a technology several known in the literature and some studies are described in the following sentences. Bellos et al. [4] compared flat plate collectors, evacuated tube collectors, compound parabolic collectors and parabolic trough collectors in a solar cooling system with a single stage absorption chiller. They finally proved that PTC is the collector which leads to the higher energetic performance. The Indian Institute of technology [5] applied this technology in a PTC of 1.5 m^2 which performance of about 50% and finally reached cooling production at 3 °C. Florides et al. [6] used TRNSYS to simulate a solar absorption system and they finally proved that 600 l of storage tank with 15 m^2 of parabolic trough collectors are able to produce the demanded cooling load for a typical building in Cyprus. Qu et al. [7] analyzed a double effect LiBr-H_2O absorption system with parabolic through solar collectors of 52 m^2 and 16 kW capacity. Simultaneously, a natural gas burner is adapted to this system in order to provide heat when solar energy is inadequate. Department of Mechanical Engineering at Guilan University [8] proved that 58 m^2 of PTC collectors are required in order to produce a cooling load of 17.5 kW.

The aim of this study is to determine the values of critical parameters for the examined system. The determination of operational parameters in dynamic simulations is something not well-established in the literature up today. The main examined parameters are the storage tank volume and the mass flow in the absorption chiller. The parabolic trough collector which is used is the commercial IST–PTC (Industrial Solar Technology Corporation product) [1]. After analyzing the system behavior in every case, a study case for a usual building is examined in order to test the system performance in a real application. The dynamic simulation of the system is the key point which makes this study interesting, because the daily operation time period of the chiller is fully determined. Moreover, the determination of the optimum specific mass flow rate and the relationship between the mass flow rate of the collector field and of the load circuit is another interesting finding. The innovative point of this work is that the parametric analysis is performed with dynamic analysis and not in steady state, as in the majority of the studies.

2. Parabolic Trough Collector (PTC) description

In this study, the parabolic trough collectors are chosen among the family of solar collectors because of their ability to operate in high temperatures (over 100 °C) and to a wide range of applications [9]. A solar power plant with PTC is studied in Ref. [10], Cabrera et al. [11] examined solar refrigeration and air-conditioning with PTC. It is obvious that PTC can be used in a great variety of applications as solar cooling, electricity production and desalination.

In this section, the heat transfer analysis inside the parabolic trough collector and the optical analysis of the sun rays are examined in order to specify thermal output of the examined collector. More specifically, the loss coefficient U_L and the optical efficiency (η_o) are presented in order to determine the collector efficiency.

The parabolic trough collector consists of a parabolic trough and a linear evacuated tube which is located in the focal line of the parabolic trough. The main idea of PTC is that the reflected radiation over the parabolic trough is directed to the focal point of parabola and so all the solar energy is concentrated in the evacuated tube. The result of this concentration is the high temperature levels in the absorber, because large amounts of energy absorbed in a small region. The use of an evacuated tube increases the thermal efficiency of the collector, because convection losses between the absorber and the cover are eliminated.

2.1. Optical analysis of PTC

The aim of this paragraph is to determine the optical efficiency of the PTC. The optical losses are taken into account by using parameters as the concentrator reflectance (ρ), the cover transmittance (τ) and the absorber absorbance (α). The other possible losses due to the design are included to the intercept factor (γ). In other words, the optical efficiency of PTC depends on the geometry of the reflector and on the material properties. All these are summarized in Eq. (1) which follows [12]:

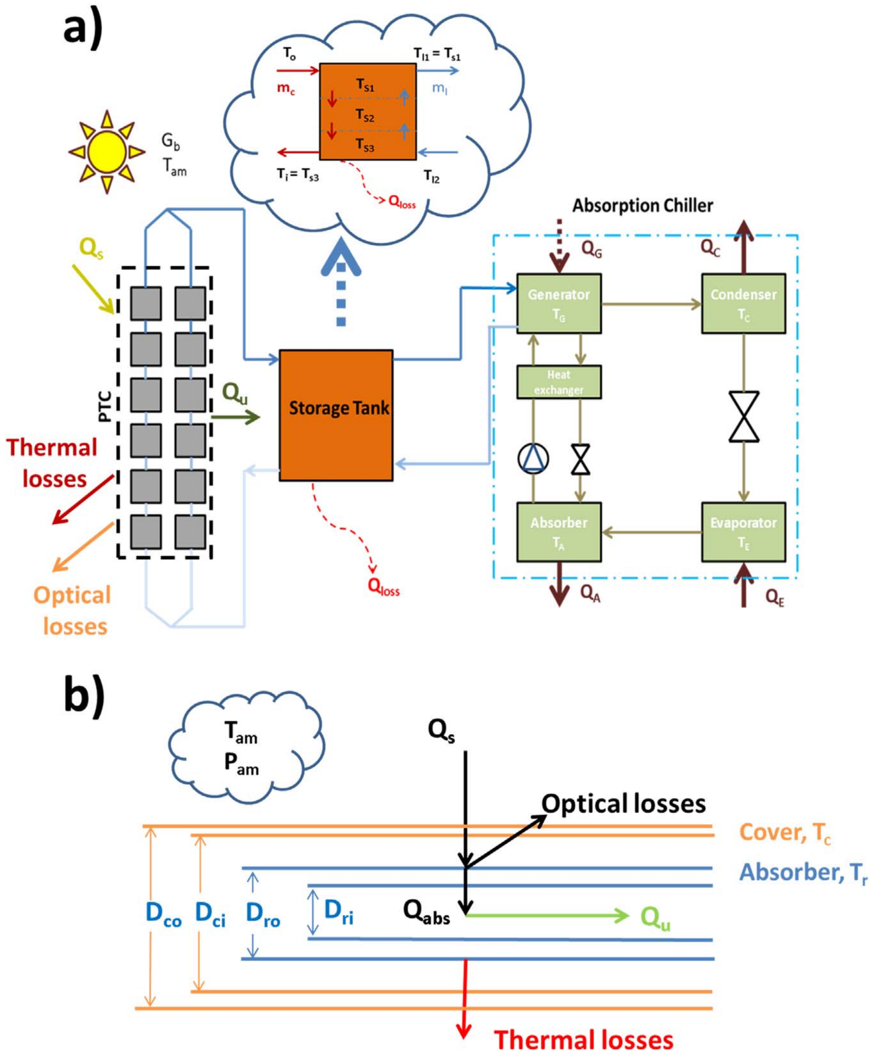

Fig. 1. (a) Solar cooling system (PTC-Storage Tank- Absorption Chiller) (b) modeling of the PTC.

$$\eta_o = \rho \cdot \tau\alpha \cdot \gamma \cdot K \tag{1}$$

The intercept factor γ is defined as the fraction of reflected radiation in the absorber to the total reflected radiation and it is selected to be equal to 0.95; a typical value according to the literature [13]. Concentration ratio (C) is defined of the aperture area to the receiver area:

$$C = \frac{A_a}{A_r} = \frac{W_a \cdot L}{\pi \cdot D_{ro} \cdot L} \tag{2}$$

A tracking system is used in the PTC in order to operate successfully. More specifically, the collector axis is located in N-S axis and the tracking system rotates the collector horizontally in E-W axis. The equation which gives the incident radiation angle during the day time is the following [12]:

$$\cos(\theta) = \sqrt{\cos^2(\theta_z) + \cos^2(\delta) \cdot \sin^2(\omega)} \tag{3}$$

The main advantage of this arrangement is that very small shadowing effects are encountered, when more than one collector is used in the first and last hours of each day.

2.2. Thermal analysis of PTC

In this section the thermal analysis of PTC is presented with the basic equations. The inner tube is the absorber and the outer the glass cover. Between these there is vacuum, approximately pressure of some Pa, fact that leads to negligible heat convection losses between absorber and cover.

The modeling of the solar collector is given in Fig. 1(b). In this study, many assumptions have been made in order to simulate the solar cooling system successfully. Below these assumptions are given with details.

Solar collector [21].

- The heat transfer fluid is incompressible
- The conduction thermal losses have not been taken into consideration
- The parabolic shape of the concentrator is symmetrical
- The solar flux is uniform in the absorber
- The ambient temperature close to the concentrator is uniform
- The effect of the shadow among the PTC modules is negligible
- The glass is opaque to infrared radiation

The useful energy gained from the collector can be determined by two ways, according to Eqs. (4) and (5) [13]:

$$Q_u = G_b \cdot \eta_o \cdot A_a - A_r \cdot U_L \cdot (T_r - T_{am}) \tag{4}$$

$$Q_u = F_R (G_b \cdot \eta_o \cdot A_a - A_r \cdot U_L \cdot (T_i - T_{am})) \tag{5}$$

It is essential to point that only the beam radiation can be exploited from the PTC because these collectors are imaging collectors with a specific image of the sun in the receiver. The thermal efficiency of collector is defined as [12]:

$$\eta = \frac{Q_u}{G_b \cdot A_a} = F_R \cdot \left[\eta_o - U_L \cdot \left(\frac{T_i - T_{am}}{G_b \cdot C}\right)\right] \tag{6}$$

The heat removal factor (F_R) which is used in the previous equation is given by the next equation [14]:

$$F_R = \frac{m_c \cdot c_p}{A_r \cdot U_L} \cdot \left[1 - \exp\left(\frac{A_r \cdot U_L F'}{m_c \cdot c_p \cdot}\right)\right], \tag{7}$$

where F' is the efficiency collector factor and can be calculated as [13]:

$$F' = \frac{1/U_L}{1/U_L + \frac{D_{ro}}{h_{fi} \cdot D_{ri}} + \frac{D_{ro}}{2k_m} \cdot \left[\ln\left(\frac{D_{ro}}{D_{ri}}\right)\right]} \tag{8}$$

The mean temperature in the receiver is given from the next equation [13]:

$$T_r = T_i + \frac{Q_u}{A_r \cdot U_L \cdot F_R}(1 - F_R) \tag{9}$$

The only parameter that is not known in the above equation is the thermal loss coefficient U_L. This parameter can be determined by Eq. (10) [13].

$$U_L = \left[\frac{A_r}{(h_{c,c-am} + h_{r,c-am})\cdot A_c} + \frac{1}{h_{r,r-c}}\right]^{-1} \tag{10}$$

The radiation coefficient between receiver and cover given as [13]:

$$h_{r,r-c} = \frac{\sigma\cdot(\mathrm{T_r}^2 + T_c^2)\cdot(T_r + T_c)}{\frac{1}{\varepsilon_r} + \left(\frac{1}{\varepsilon_c} - 1\right)\cdot\left(\frac{A_r}{A_c}\right)} \tag{11}$$

The Stefan-Boltzmann constant (σ) is equal to $5.67\cdot10^{-8}$ W/m^{2}K^4. The other radiation coefficient between the cover and the sky is presented in the next equation [13]:

$$h_{r,c-am} = \varepsilon_c\cdot\sigma\cdot(\mathrm{T_c}^2 + T_{sky}^2)\cdot(T_c + T_{sky}) \tag{12}$$

The sky temperature (T_{sky}) is considered to be 6 degrees less than ambient temperature [15]. The convection coefficient for the water inside the tube (h_{fi}) is calculated from heat transfer theory [16]. Firstly the Nusselt number is calculated and after h_{fi} according to the following equations. It is important to state that the flow is turbulent (Re > 2300) [17] and for this reason the Eq. (13) is used.

$$Nu = 0.023\cdot\mathrm{Re}^{0.8}\cdot\mathrm{Pr}^{0.4} \tag{13}$$

$$Nu = \frac{h_{fi}\cdot D_{ri}}{k_{fi}} \tag{14}$$

Now, the convection coefficient between cover and air is calculated with the next equation [13]:

$$h_{c,c-am} = \frac{8.6\cdot V_{air}^{0.6}}{L^{0.4}} \tag{15}$$

According to literature, if this equation gives value less than 5 W/m^2 K, then the coefficient takes the value of 5 W/m^2 K. The specific mass flow rate in the collector can be determined as:

$$m_a = \frac{m_c}{A_a} \tag{16}$$

Finally, it is essential to say that all these equations are solved together and iteratively in a FORTRAN program. The methodology of the way that the equations have been solved has been presented by Tzivanidis et al. [18–20]. It is also important to state that a lot of information has also taken by the developed numerical model of Ghodbane et al. [21–25].

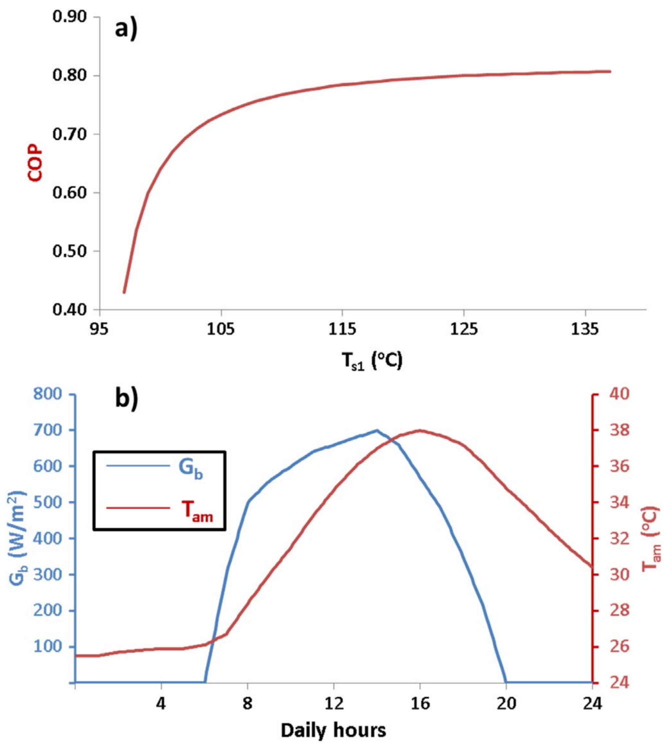

Fig. 2. (a) COP for various load temperature levels (b) Weather data, left axis shows the radiation (blue line) and right axis the ambient temperature (red line). (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

3. System description

In this paragraph, the examined system is presented analytically. The main parts of this system are the parabolic trough collectors, the storage tank and the absorption chiller (Fig. 1(a)). The water enters to the solar field and its temperature increases because the solar energy is captured by the collectors. The hot water enters into the upper part of the storage tank and heats the stored water. In the other side of the tank, hot water lets the tank and goes to the generator of the chiller in order to give the demanded heat. After leaving the generator, it returns to the down part of the tank with a lower temperature. The cooling load is produced in the evaporator of the chiller and this is the output of the examined system. It is important to state that the storage tank losses energy to the environment due to its high temperature, something that is taken into consideration in the presented study. In the following parts of this paragraph the separated systems will be presented except from PTC which is presented in Section 2.

3.1. Storage tank modeling

In the system, a storage tank for hot water is used in order to have a water supply the hours after sunset. The design of it is very important for the system, because it determines the daily operation time of the system and simultaneously influences on temperature level in the system. Using a large tank, the water storage increases but the mean water temperature is lessening which leads to lower chiller performance and by using a small tank the results are versus. Thus, an optimization of the storage tank volume is needed in order to select the parameters which lead to the desired performance. It is essential to state that the water inside the tank is pressurized in order to be kept in liquid phase for temperature level over 100 °C. More specifically, the pressure is selected to be close to 8 bars which lead to no steam creation because the saturation temperature is equal to 170 °C in this pressure level. In the present modeling, the tank was separated in three thermal zones (over, medium and down) in order to simulate the temperature distribution inside it [4]. The high part is warmer because the hotter water has lower density and moves to the upper part. For this reason, the hot water from solar collectors enters into the upper part of the tank while cold water from the down part is used in the inlet of the collector field. Fig. 1(a) includes the storage tank modeling and the way that it is analyzed in this study.

The energy balance equations of the storage tank mixing zones are given below [13]:

$$M_{S1}\cdot c_p\cdot\frac{\partial T_{S1}}{\partial t} = m_C\cdot c_p\cdot(T_o - T_{S1}) + m_L\cdot c_p\cdot(T_{S2} - T_{S1}) - U_T\cdot(A_{T,S3})\cdot(T_{S1} - T_{am}), \tag{17}$$

$$M_{S2}\cdot c_p\cdot\frac{\partial T_{S2}}{\partial t} = m_C\cdot c_p\cdot(T_{S1} - T_{S2}) + m_L\cdot c_p\cdot(T_{S3} - T_{S2}) - U_T\cdot(A_{T,S2})\cdot(T_{S2} - T_{am}), \tag{18}$$

$$M_{S3}\cdot c_p\cdot\frac{\partial T_{S3}}{\partial t} = m_C\cdot c_p\cdot(T_{S2} - T_{S3}) + m_L\cdot c_p\cdot(T_{L2} - T_{S3}) - U_T\cdot(A_{T,S3})\cdot(T_{S3} - T_{am}), \tag{19}$$

These differential equations are solved together because the system is complex and the temperature of the one part influences the temperature of the other.

3.2. Absorption chiller modeling and other parameters

A single-effect absorption chiller operating with the working couple LiBr-H_2O is the used sorption machine. The assumptions of **absorption chiller are:**

- The evaporator temperature is equal to 10 °C
- The outlet temperature of the hot stream in the generator is 7 K lower than the inlet temperature

This system is powered by a parabolic trough collector module (PTC) in order to produce refrigeration at 10 °C; an adequate temperature for cooling applications. The COP is taken by the literature [4] and it is given as a function of the load temperature (T_{s1}) in Fig. (2a). It is essential to state that temperature difference between the load streams (inlet and outlet) are given by Eq. **(20)** [4]:

$$T_{S2} = T_{S1} - 7, \tag{20}$$

The heat input in the chiller is given by equation 24:

Table1
Engineering parameters [1,25].

Parameter	Value	Parameter	Value
Tube length (L)	6.1 m	Specific tube flow rate (m_a)	0.03 kg/(sm^2)
Parabola width (W_a)	2.3 m	Nodes per hour	300
Focal distance (f)	0.8 m	Tank loss coefficient (U_T)	0.7 W/m^{2}K
Receiver outer diameter (D_{ro})	51 mm	Cover outer diameter (D_{co})	74 mm
Receiver inside diameter (D_{ri})	47 mm	Cover inside diameter (D_{ci})	70 mm

Table 2
Optical properties [1,25].

Parameter	Value
Concentration ratio (C)	14.4
Cover emittance (ε_c)	0.88
Absorber emittance (ε_r)	0.10
Trough reflectance (ρ)	0.90
Receiver absorbance (α)	0.95
Cover transmittance (τ)	0.85
Cover emittance (ε_c)	0.88

$$q_L = m_L \cdot c_p \cdot (T_{L1} - T_{L2}), \tag{21}$$

The produced cooling is calculated with the COP, as Eq. **(22)** shows:

$$q = COP \cdot q_L, \tag{22}$$

The weather data for solar beam radiation and for the ambient temperature distributions were taken from TRNSYS for a typical day of July in Athens. Fig. (2b) shows the weather data used in this analysis. A very hot day was selected in order to examine the system in the most difficult ambient conditions. The cooling systems face difficulties the days with higher temperature levels, something that is taken into account with this choice. The time axis in Fig. (2b) corresponds to local time of Athens. Moreover, the ambient pressure was selected to be equal to 1 bar.

It is important to state the absorption chiller operates better when the generator temperature and the evaporator temperature are greater. On the other hand, the system performs better for lower absorber and condenser temperature levels.

The dimensions of PTC are taken from a real collector (PT1-IST Company) [1,25] and the other parameters were selected to have typical values according to respect the bibliography [26–31]. The engineering properties are given in Table 1 and the optical properties in Table 2.

3.3. Methodology of the study

The general methodology of this study is depicted in Fig. 3. The first step is the selection of the examined parameters. These parameters and the weather data are the basic inputs of the developed program. Then the numerical model is applied for all the times steps and after integration over the day is made in order to take the final results. It is important to be stated that in the numerical model, are the used equations have been given in the previous sections. The analysis is performed for a typical summer day in Athens.

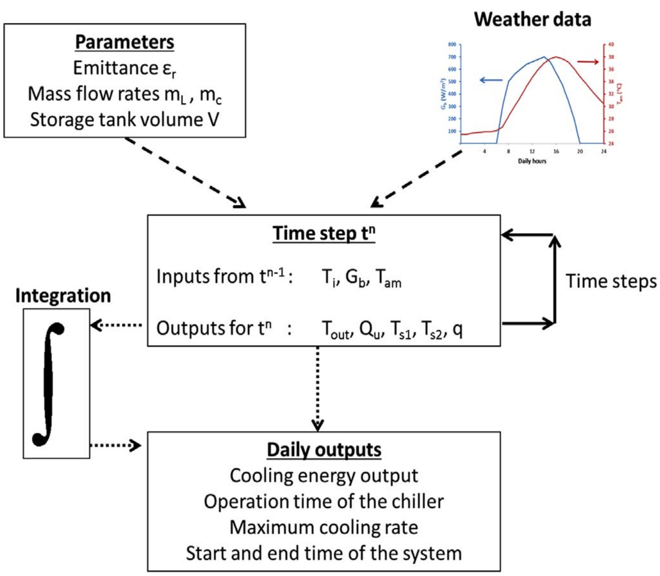

Fig. 3. Flow chart of the methodology.

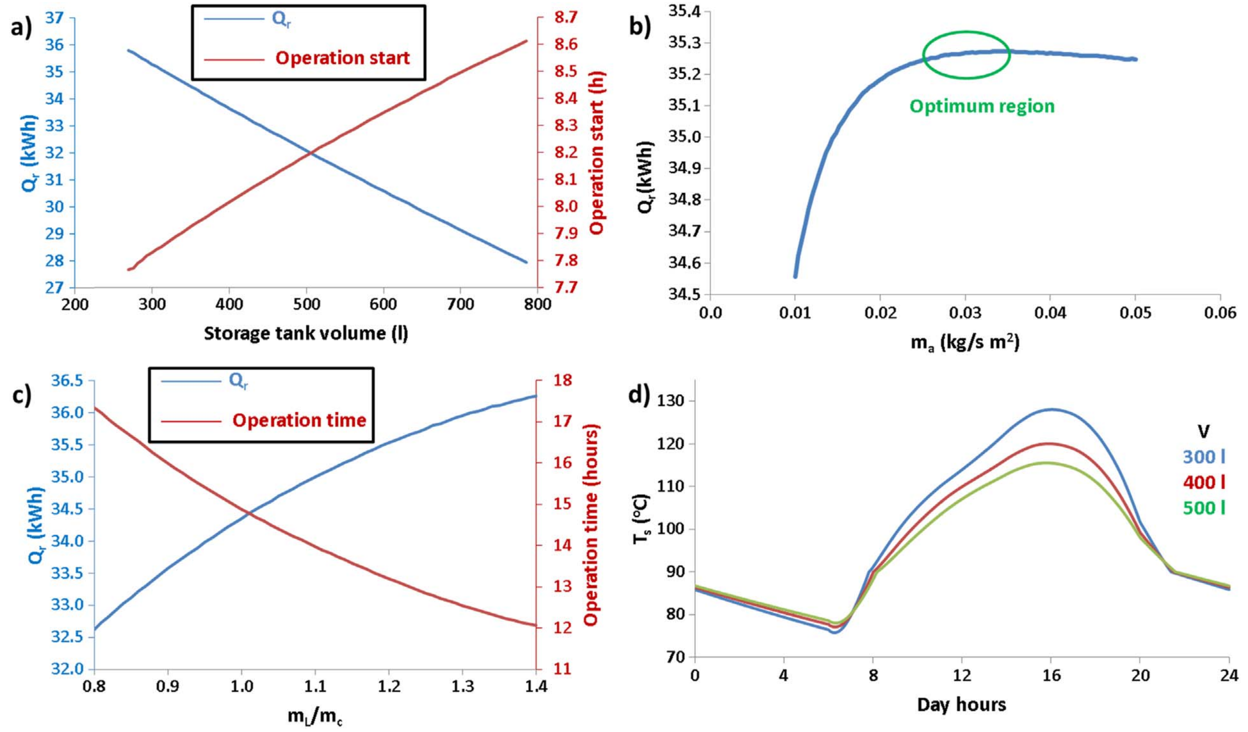

Fig. 4. (a) Mass fraction influence on system performance (b) System performance with specific mass rate in collector loop (c) Tank volume influence on system performance (d) Temperature inside the tank for different tank volumes.

4. Parametric results and discussion

4.1. Mass flow rate impact on the performance

The mass flow rate in the absorption chiller should be determined from Fig. (4a). Different values for the mass rate are investigated and in every case the daily refrigeration output is calculated. It is essential to state that the horizontal axis is the ratio of mass flow rate in the load circuit (from tank to the chiller) to the mass flow rate to the collector circuit. In this parametric analysis, the storage tank volume was selected to be equal to 300 [L]. This value is the most suitable, as it will be determined in the nest paragraphs.

Fig. (4a) shows that the increase of mass to absorption chiller gives greater refrigeration but leads to less operation hours, so the optimum solution is in the middle of the diagram. In order to operate the system for many hours per day and to have enough cooling load, the ratio m_L/m_c was selected to be equal to 1.15. In other words, the load mass flow rate will be 15% greater than the mass flow rate in the collector system. The next step is to determine the mass flow rate in the collector loop. Fig. (4b) exhibits the way that the cooling load varies when the specific mass flow rate in the collector loop takes different values. The mass flow in the chiller was the same in all cases. The optimum performance is achieved for the specific mass rate of 0.03 kg/sm^2 approximately. Thus, this value is

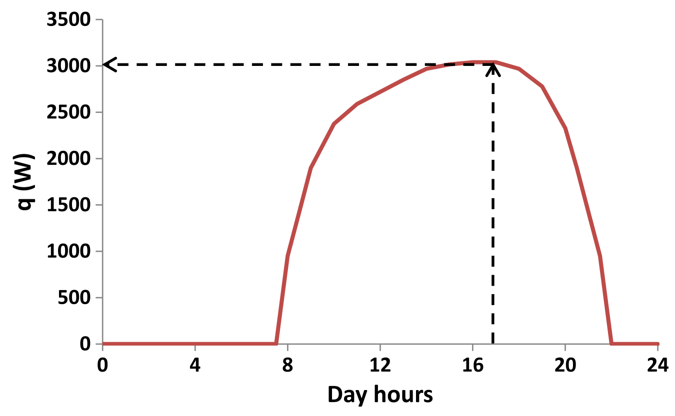

Fig. 5. Daily system performance.

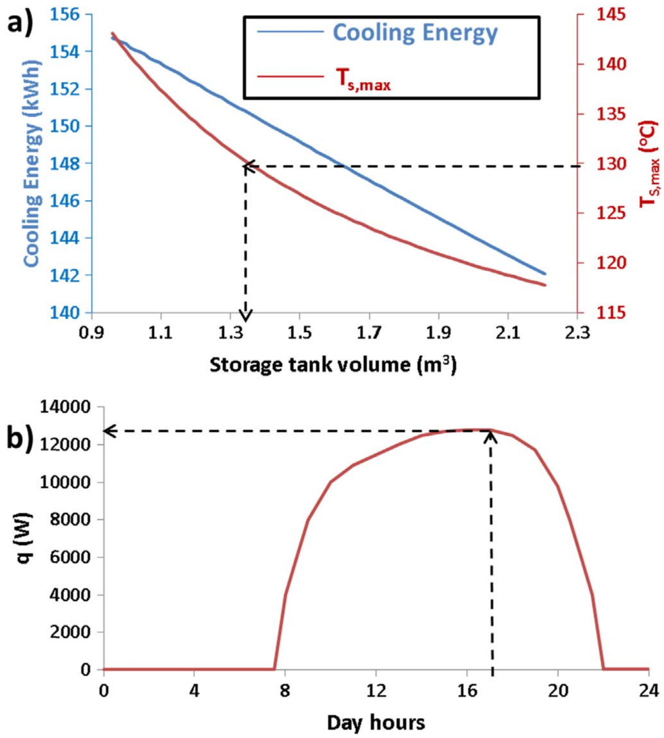

Fig. 6. (a) Tank volume selection for the building case (b) System performance in the building case.

selected as the optimum and it is used in the following analysis.

4.2. Impact of storage tank volume on the performance

The next parameter that influences on the system performance is the storage tank volume because it determines the quantity of stored water and the mean operating temperature level. More specifically, a large tank stores more hot water and the system is able to operate some hours after the sunset but the chiller starts lately in the morning. On the other hand, a small storage tank leads to high temperatures inside with a quick start in the morning but the operation will stop earlier in the night. Fig. 4(c) illustrates the impact of storage tank volume in the daily system performance.

According to Fig. 4(c), a small tank is the optimum solution for the examined system, because it leads to greater refrigeration. Also the chiller starts the operation earlier in the morning, because the water mass in the tank is lower and this reaches the preferable temperature rapidly. In real applications, the need of refrigeration starts before 8:00 in the morning (for commercial buildings), so it was selected a tank with 300 [L] in order to fulfill this restriction and to have a great amount of cooling.

Temperature level inside the tank is also a very important for the system analysis, because a great temperature leads to huge losses to the environment and a very low temperature reduces the chiller performance. Fig. 4(d) shows the daily profile of the mean temperature in the upper part of the tank for 3 different storage tank volumes.

According to Fig. 4(a), the maximum temperature inside the tank is observed lately in the noon when the ambient temperature is maximized (about 16:00). Moreover, the lower tank of 300 [L] has greater temperature during the day because has lower thermal capacity and it warms up easily. On the other hand, this tank stores lower heat quantity and after the sunset its temperature is lower than the other cases. The maximum temperature in the case of 300 [L] is about 130 °C; a value which will be used in the next paragraph as selection criterion. This analysis explains the role of the storage tank in the system and the impact of the volume in the temperature profile during the day.

4.3. Cooling load profile

In this point, the cooling load during the day will be presented using the data from above analysis (tank volume 300 [L] and mass load flow 15% greater than collector mass flow). Fig. 5 depicts the cooling load and simultaneously the operation hours.

It is shown that the maximum cooling load is about 3 kW and the system operates close to this load for about 8 h daily. The total operation time is 13 h with a total cooling energy of 33.5 kW h, with the daily solar beam radiation potential to be 95.3 kWh. The ratio of these quantities gives the system performance to be 0.352 and it is about the half of chiller COP. The reasons are the existence of thermal losses in the solar collectors, the thermal losses in the tank and the thermal inertia of the tank in the morning. The produced cooling load of about 3 kW is able to cool a room with an area of 25 m^2 approximately.

5. A study case for a building

In this section, the performance of a solar cooling system for a building of 100 m^2 in Athens will be examined. From the previous analysis it is found that with one collector it is possible to refrigerate successfully a room of 25 m^2. For the larger building the use of more collectors in parallel connection will be analyzed in order to find the optimum solution and simultaneously the optimum tank volume will be determined again (Fig. 6(a)). It is known that for area 25 m^2 a chiller with capacity 3 kW is needed [32], so in this case 12 kW is the needed capacity and consequently 4 parabolic collector modules will be used in order to cover the maximum cooling load. The specific mass flow rate was selected to be 0.03 kg/sm^2, but the storage tank volume has to be investigated again. Fig. 6(a) shows the cooling energy production and the mean tank temperature for various values of the storage tank volume.

For normal operation of system, the maximum temperature inside the tank should be lower than 130 °C [4], so a volume of 1.35 m^3 is selected for the building. This is in accordance with the optimization of the storage tank volume in the previous paragraph for the case of one PTC module. Greater storage tank volume leads to lower cooling energy production because of the reduction in the operation time of the chiller. Specifically, a greater tank needs more energy to start its operation, so the system starts to operate lately in the morning. Fig. (6b) exhibits the system cooling output during the day.

The system is able to give 150 kW h with a total operation time 12.5 h. The load is about 12.8 kW and it is produced at 17:00. This result is explained by the time lag because of the storage system.

Moreover, according to Fig. 6(a), greater storage tank volume leads to lower mean temperature inside the storage tank and to lower cooling capacity. These results are logical because lower temperature levels lead the absorption chiller to have a lower COP and the produced cooling is lower. According to Fig. (6b), the system starts to operate about 7:30 in the morning. Up to 9:00, the produced cooling load increases with a high rate due to the increase of the available solar energy. After this point, the increase has a lower rate and the maximum cooling production is observed at 17:00. After this time, the solar potential is lower and the cooling production is getting lower up to 22:00, when the system stops operating. These are logical results because the system operates some ours after sunset due to the storage tank.

6. Conclusions

In this study, a solar refrigeration system powered by parabolic trough collectors is analyzed parametrically. The innovative of this study is the determination of crucial parameters by studying the system in dynamic state, using real weather data for Athens. There is no other energy source, fact that makes this system fully renewable. The system is parametrically analyzed firstly and after a case study for a building in Athens is given. The main conclusions are summarized in the following sentences:

- Greater mass flow rate in the absorber chiller leads to higher refrigeration rate but the total operation time reduces. Thus, the choice of this parameter depends on the examined application and the operation hours.
- The specific mass flow rate in the solar collector loop influences on the system performance and the optimum value is about 0.03 kg/sm^2. This result is important because it is proves the need of mass flow rate optimization in every solar cooling system.
- The storage tank volume is a critical parameter because it influences on the cooling and on the operation time. A large volume leads to lower performance but the operation stops lately after sunset. On the other hand, a small volume creates a system that starts earlier in the morning and performs very well but stops operating earlier after sunset and the temperatures inside the tank are high.
- One module of PTC coupled with a tank of 300 [L] is able to give 3 kW cooling load with daily solar energy 95.3 kWh which means a mean solar coefficient of performance equal to 0.352, an accepted value.
- By using 54 m^2 collecting area and a storage tank of 1.35 m^3 it is possible to success a maximum cooling load of 12.8 kW, which means that a building of 100 m^2 is cooled with an adequate way.

Finally, it is proved that this system is able to operate successfully and to produce sufficient cooling rate during a sunny and hot day in Athens.

Acknowledgements

Evangelos Bellos would like to thank Onassis foundation for its financial support.

References

[1] E. Bellos, C. Tzivanidis, K.A. Antonopoulos, G. Gkinis, Thermal enhancement of solar parabolic trough collectors by using nanofluids and converging-diverging absorber tube, Renew. Energy 94 (94) (2016) 213–222.
[2] H. Majdi, Performance evaluation of combined ejector LiBr/H2O absorption cooling cycle, Case Stud. Therm. Eng. 7 (2016) 25–35.
[3] D.A. Kouremenos, K.A. Antonopoulos, E.S. Domazakis, Solar radiation correlations for the Athens, Greece, area, Sol. Energy 35 (3) (1985) 259–269.
[4] E. Bellos, C. Tzivanidis, K.A. Antonopoulos, Exergetic, energetic and financial evaluation of a solar driven absorption cooling system with various collector types, Appl. Therm. Eng. 102 (2016) 749–759.
[5] H. Raheman, C.P. Gupta, Development of a solar-energy-operated vapor- absorption-type refrigerator, Appl. Energy 34 (1989) 89–98.
[6] G.A. Florides, S.A. Kalogirou, S.A. Tassou, L.C. Wrobel, Modelling and simulation of an absorption solar cooling system for Cyprus, Sol. Energy 72 (1) (2002) 43–51.
[7] M. Qu, H. Yin, D.H. Archer, A solar thermal cooling and heating system for a building: experimental and model based performance analysis and design, Sol.

Energy 84 (2) (2010) 166–182.
[8] Mazloumi, M., Naghashzadegan, M., Javaherdeh, K., Simulation of a solar absorption cooling system with parabolic trough collector for sunshine hours, in: Proceedings of the 47th conference on simulation and modeling. Helsinki, Finland, SIMS, 2006,pp. 188–193
[9] A. Valan Arasu, T. Sornakumar, Design. manufacture and testing of fiberglass reinforced parabola trough for parabolic trough solar collectors, Sol. Energy 81 (10) (2007) 1273–1279.
[10] C. Tzivanidis, E. Bellos, K.A. Antonopoulos, Energetic and financial investigation of a stand-alone solar-thermal Organic Rankine Cycle power plant, Energy Convers. Manag. 126 (2016) 421–433.
[11] F.J. Cabrera, A. Fernández-García, R.M.P. Silva, M. Pérez-García, Use of parabolic trough solar collectors for solar refrigeration and air-conditioning applications, Renew. Sustain. Energy Rev. 20 (2013) 103–118.
[12] S. Kalogirou, Solar thermal collectors and applications, Prog. Energy Combust. Sci. 30 (3) (2004) 231–295.
[13] J.A. Duffie, W.A. Beckman, Solar engineering of thermal processes, third ed., Wiley, Hoboken, NJ, USA, 2006.
[14] S. Kalogirou, A detailed thermal model of a parabolic trough collector receiver, Energy 48 (1) (2012) 298–306.
[15] H. Gaven, R. Bannerot, Optical and thermal analysis of parabolic trough solar collectors for technically less developed countries, Houston, Texas, 1984.
[16] V., J. Lienhard I, V., J. Lienhard, A Heat Transfer Textbook, Fourth edition, Phlogiston Press, USA, 2012.
[17] Bellos E., Tzivanidis C., Korres D., Antonopoulos K.A. Thermal analysis of a flat plate collector with Solidworks and determination of convection heat coefficient between water and absorber. in: ECOS conference 2015, Pau, France
[18] C. Tzivanidis, E. Bellos, D. Korres, K.A. Antonopoulos, G. Mitsopoulos, Thermal and optical efficiency investigation of a parabolic trough collector, Case Stud. Therm. Eng. 6 (2015) 226–237.
[19] E. Bellos, D. Korres, C. Tzivanidis, K.A. Antonopoulos, Design, simulation and optimization of a compound parabolic collector, Sustain. Energy Technol. Assess. 16 (2016) 53–63.
[20] S.R. Pavlović, E. Bellos, V.P. Stefanović, C. Tzivanidis, Z.M. Stamenković, Design, simulation and optimization of a solar dish collector spiral-coil thermal absorber, Therm. Sci. 20 (4) (2016) 1387–1397.
[21] M. Ghodbane, B. Boumeddane, A numerical analysis of the energy behavior of a parabolic trough concentrator, J. Fundam. Appl. Sci. 8 (3) (2016) 671–691.
[22] M.Ghodbane, B.Boumeddane, S.Largot, N.E.H.Heniat, Etude optique et thermique d′un concentrateur cylindro-parabolique en site d′alger, algerie, presented at IXth International Congress on Renewable Energy and the Environment, Djerba, Tunisie, 18–20 March, 2015.
[23] M. Ghodbane, B. Boumeddane, Numerical modeling of a parabolic trough solar collector at bouzaréah, Algeria, Int. J. Chem. Pet. Sci. 4 (2015) 11–25.
[24] M. Ghodbane, B. Boumeddane, S. Largot, NE.H. Heniat, Simulation numérique d'un concentrateur cylindro-parabolique en el oued, Algérie, Int. J. Sci. Res. Eng. Technol. (IJSET) 3 (2015) 68–74.
[25] A. Fernández-García, E. Zarza, L. Valenzuela, M. Pérez, Parabolic-trough solar collectors and their applications, Renew. Sustain. Energy Rev. 14 (7) (2010) 1695–1721.
[26] N. Eskin, Transient performance analysis of cylindrical parabolic concentrating collectors and comparison with experimental results, Energy Convers. Manag. 40 (2) (1999) 175–191.
[27] ⟨http://www.schott.com⟩.
[28] E. Bellos, C. Tzivanidis, K. Moschos, K.A. Antonopoulos, Energetic and financial evaluation of solar assisted heat pump space heating systems, Energy Convers. Manag. 15 (2016) 306–319.
[29] C. Tzivanidis, E. Bellos, G. Mitsopoulos, K.A. Antonopoulos, A. Delis, Energetic and financial evaluation of a solar assisted heat pump heating system with other usual heating systems in Athens, Appl. Therm. Eng. 106 (2016) 87–97.
[30] E.Bellos, C.Tzivanidis, A.Delis, K.A.Antonopoulos, Comparison of a solar driven heat pump heating system with other typical heating systems with TRNSYS, in: Proceedings of the 28th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems, 30th June – 3rd July 2015 Pau, France
[31] G. Mokhtar, B. Boussad, S. Noureddine, A linear Fresnel reflector as a solar system for heating water: theoretical and experimental study, Case Stud. Therm. Eng. 8 (2016) 176–186.
[32] ⟨http://inventoraircondition.gr⟩.